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An in-arm suspension unit was developed for retrofit onto a Marine Corps Assault Amphibian Vehicle. The hydropneumatic system utilizes notrogen gas for springing and multiple wet brake plates for dampening. The system offers twelve inch jounce, five inch rebound, and 3.5g loading for a static load of 6200 pounds.

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DAVID TAYLOR NAVAL SHIP
R & D CENTER

DESIGN REPORT
FOR AN
IN-ARM HYDROPNEUMATIC
SUSPENSION UNIT

MODEL GKA1-AAV-01-03

SEPTEMBER 1988

DAVID TAYLOR

CONTRACT NO. N00167-88-C-0024

DATED 4 JANUARY 1988

Cadillac Gage TEXTRON
Cadillac Gage / Subsidiary of Textron Inc.

CADILLAC GAGE

89 2 16 159

FINAL DESIGN REPORT

IN-ARM SUSPENSION

MODEL NUMBER 6KA1-AAV-0103

(CDRL A010)

PREPARED BY: Susan M. Stack DATE: 9/1/88

APPROVED: James J Fibbs DATE: 9/1/88

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1.0 INTRODUCTION

This report describes the design and analyses performed to complete Task I, (Contract #N00167-88-C-0024), the design of a Hydropneumatic Suspension System for the AAV-7A1 amphibian assault vehicle.

The Cadillac Gage Textron In-Arm Suspension Unit (ISU) is designed to provide a superior alternative to the current torsion bar system by offering improved ride quality, reduction in track tension variation, and negligible vehicle interior volume necessary for suspension installation. These improvements result in higher vehicle speeds through difficult terrain, lower track wear due to more consistent track tension and road wheel/track contact, and increased interior vehicle volume.

1.1 Background and Scope

The award of Contract #N00167-88-C-0024 in January 1988 by the Marine Corps Program Office of the David Taylor Naval Ship Research and Development Center (DTNSRDC) to Cadillac Gage Textron Inc. began Task I of a four-task program, which in addition to the initial design phase, includes fabrication and testing (Task II), vehicle installation, instrumentation and field test support (Task III), and the final report (Task IV), as is specified in the Statement of Work (SOW) (see Appendix A). The suspension system shall consist of 12 hydropneumatic suspension units and 2 spare units.

1.2 Objective

The objective of Task I was to design a hydropneumatic ISU and system meeting DTRC's design criteria and fulfilling their requirements for a system that improves ride performance and requires less interior volume than the current torsion bar system at a reasonable cost. This objective has been met with the design presented in the following pages.

2.0 SUSPENSION DESIGN SUMMARY

2.1 System and Unit Requirements

The final ISU configuration was arrived at based on allowable space claim, load requirements, and contract design criteria, as well as results from classical stress, finite element, weight, and heat transfer analysis of critical components, and computer mobility modeling (VEHDYN) of the vehicle. Table I lists the design parameters for the proposed and final design of the suspension unit and system.

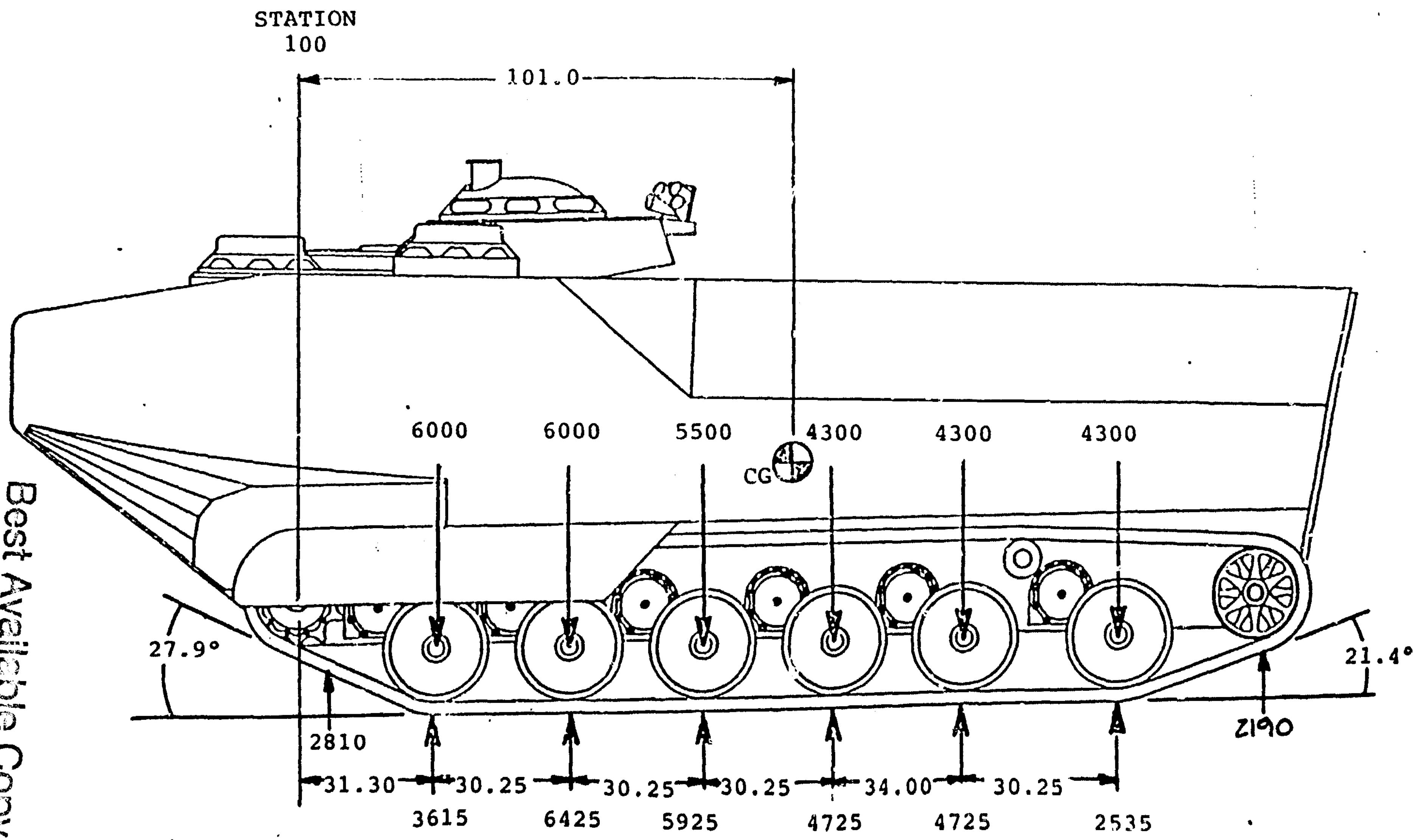
2.2 Vehicle Load Requirements

The road wheel loading requirements for the vehicle were calculated using the gross vehicle weight (GVW) and dividing by ten road wheels (assuming 50 percent of the front and rear stations is taken up by track tension). This gives an average load per road wheel. Because the vehicle is known to be front-heavy, more of the distributed load was placed at stations 1 and 2 for balance. The track tension was approximated at 10 percent of the GVW per side, or 6,000 lbs. By summing the forces and moments to zero, the desired load distribution was determined, with the average static road wheel load being 6,000 lbs; at 15 percent growth, this load is 7,200 lbs. Subsequent calculations were based on these values. The ground loads at the front and rear stations were found by subtracting the product of the track tension and the sines of the entry and departure angles from the road wheel loads at those stations, respectively. The road wheel locations were established by taking into consideration the space claim of the mounting bracket for each unit, possible interference between the unit and the road wheel, and the placement of the support rollers. Figure 1 shows the load distribution and system dimensions.

2.3 Suspension Description

For discussion purposes, the unit can be separated into three basic functional elements. These are:

STATIC ROAD WHEEL LOAD CHARACTERISTICS



GVW = 55,865 LBS
SPRUNG WEIGHT = 50,765 LBS
[illegible] 6,000 LBS

Best Available Copy

Figure 1

TABLE I

Design Requirements Summary

Suspension System	CGTI Proposed Design	CGTI Final Design	Ref. Para.
Weight (lbs)			
Unit	256.9	282.2	2.8
System	3330.0	3718.4	3.1
Pitch Frequency (Hz.)	0.77	0.882	3.0
Bounce Frequency (Hz.)	1.10	1.009	
Temperature Effects (-65 °F to 125 °F)	1" vertical Change per 60°F	Same as proposed	2.4.1.1
Vehicle Weight Growth (%)	15	15	2.3.1.1
Durability (hrs to major rebuild)	1,000	1,000	-
Maintenance	100 Dynamic Hours	Same as proposed	2.7
Corrosion Protection	Report included	Same as proposed	2.4.2
Ballistic Protection	7.62-mm AP @ muzzle and 0° obliquity (2)	Same as proposed	2.5
Installation			
Lateral Clearance (in.)			
Track centerline to hull	12.625	12.625	
Track to hull	2.125	2.125	
Ground Clearance (in.)	16	18	3.0
Track Length on Ground (in.)	155	155	-
Suspension Unit			
Structural			2.3.3.2
Bearing Loads (lbs)			
Vertical	100,000	34,245	
Lateral	20,000	20,000	
Combined			
Vertical	45,500	21,600	
Lateral	17,300	17,300	

TABLE I

Design Requirements Summary

(Continued)

Suspension System	CGTI Proposed Design	CGTI Final Design	Ref. Para.
Spring System			2.3.1
Road Wheel Travel (in.)			
Jounce Stations 1 & 2	10.0	12.5	
Jounce Stations 3-6	12.0	12.5	
Rebound, all stations	4.0	5.0	
Spring Rate (g's) max.	3.5	3.5 at 6K 3.0 at 7.2K	
Damper System			
Damper Rate (lbs) max.	5,000	6,000	2.3.2

- Spring - The spring system provides the necessary resistive force by compressing a nitrogen charge to oppose road wheel forces input by static vehicle load and terrain disturbances over which the vehicle is traveling.

- Damper - The damper system provides necessary forces to minimize vehicle pitch and heave and is activated in response to the velocity and position of the road wheel. The damper is comprised of a hydromechanical, wet multiple-friction disc mechanism located within the center section of the roadarm spindle.

- Structure - The ISU structure contains the damper spring and bearing systems and is designed to meet or exceed the load criteria as required by the vehicle application and/or the customer specification.

The analytical determination of specific values cited in these sections appears in Appendix B.

2.3.1 Spring System.

2.3.1.1 System description.

The hydropneumatic in-arm suspension unit spring is a slider-crank mechanism, as depicted in Figure 2. Rotation of the roadarm in the jounce direction causes an increase in the compression of the entrapped nitrogen volume. This compression results in an increase in gas temperature and pressure as described by the nitrogen real gas equations. An increase in pressure is translated by the slider-crank linkage and results in increasing torque on the roadarm. Thus interaction of geometry and gas compression forms the gas spring system.

Several key elements comprise the spring system. Variations of these parameters within specified design constraints result in changes to the

GAS SPRING

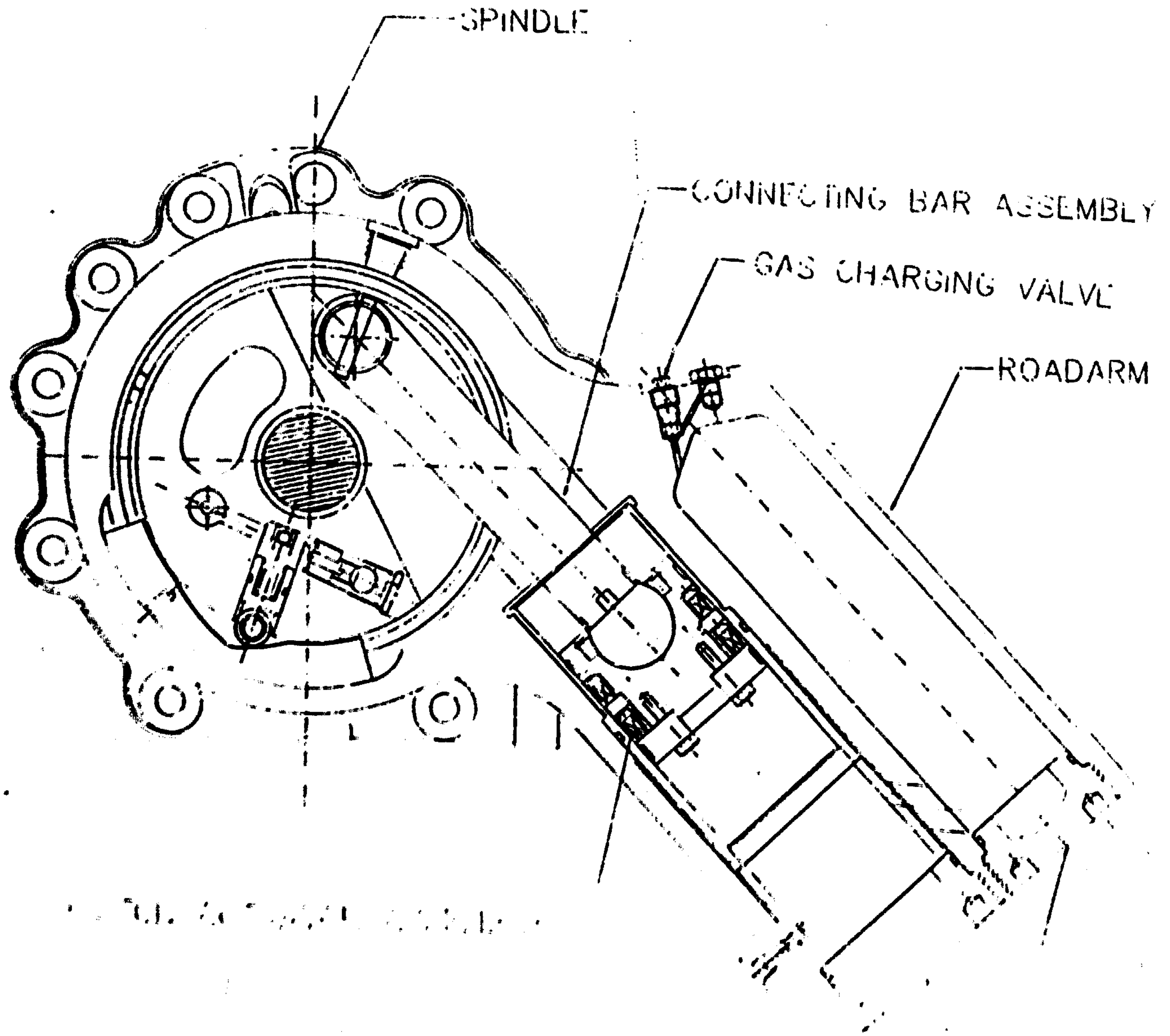


Figure 2

overall gas spring performance. These changes are manifested by the road wheel position versus force (spring) curve. These elements are:

- Slider-crank geometry
- Entrapped gas volume/oil volume
- Gas precharge pressure

The following design constraints were dictated by the vehicle interface requirements and customer desires:

- Minimum static wheel load 6,000 lbs
- 15 percent growth static load
- Roadarm length 16 inches
- Lateral clearance 0.38 inches (side wall to wheel)
- Maximum static pressure 3,500 psig
- Gas spring leakage rate - less than one inch ride height change in six months, 1,500 miles, or 150 operating hours.

2.3.1.2 <u>System spring rate</u>.

The spring system produces a spring rate which varies as a function of wheel position. The rate at the static wheel position is relatively low, providing improved ride quality. However, as the wheel approaches the jounce position, the spring rate increases dramatically, to prevent "bottoming out" on rugged terrain. The characteristics of this variable rate spring are shown in Figure 3.

The lower initial spring rate of the Cadillac Gage unit retains a significant proportion of the static wheel force into the rebound position. This load maintains good contact between the track and road wheels which reduces the variation in track tension and decreases the probability of misguides and track throwing. In addition, the lower initial spring rate permits operating with reduced track tension which lowers the vehicle's rolling resistance, thus decreasing fuel consumption, reducing wear on the running gear, and minimizing dynamic forces resulting from high speed crossings of large obstacles.

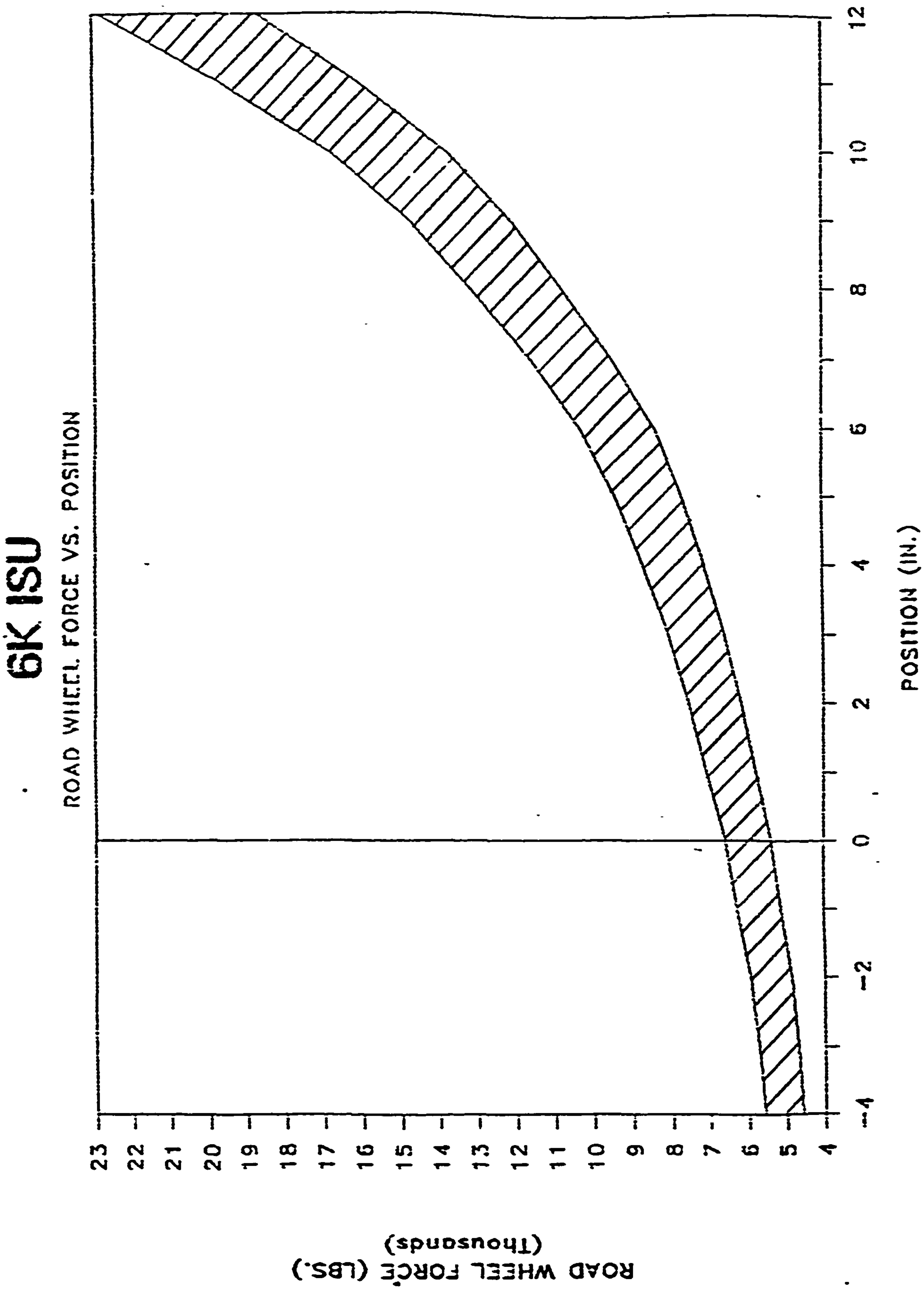


Figure 3

Road Wheel Load vs. Position

2.3.1.3 Spring geometry.

The gas spring crank geometry was established by incorporating many of the design parameters specified in paragraph 2.3.1.1; i.e., roadarm length, lateral clearances, maximum static pressure, and road wheel stroke.

The lateral space claim was the first to be evaluated to determine the maximum gas piston diameters which could be installed in the roadarm. To determine the roadarm width, nominal clearances of 0.38 inches from wheel to roadarm and 0.38 inches from roadarm to hull were subtracted from the lateral space claim. A view of this space claim is shown in Figure 4. Mounting bracket and wheel spindle thicknesses were also taken into consideration. Next, because the roadarm is also a structural member, a maximum gas pressure and input road wheel load evaluation was made to determine the minimum wall required. Preliminary stress calculations and past experience with a similarly sized model dictated this to be 0.5 inches.

The conclusion of this analysis is that the optimum piston diameter which provides an acceptable structural safety margin and fits the envelope is 3.50 inches.

To optimize the design, a 16-inch roadarm length was chosen. The effective roadarm length dictates the spring reaction torque required to balance the input road wheel forces at the static position, as well as the angular travel required to achieve the desired rebound and jounce distances. With 6,000 pounds road wheel load at the static road wheel position, the torque is 77,780 in-lbs. With a vehicle growth of 15 percent and a static road wheel load of 7,200 pounds, the required torque is 93,300 in-lbs.

The connecting bar pin location was established to minimize the piston side load at the static road wheel position and also determined the minimum inside roadarm bearing diameter, which was found to be 8.3125 inches.

VEHICLE INSTALLATION WITH MOUNTING BRACKET

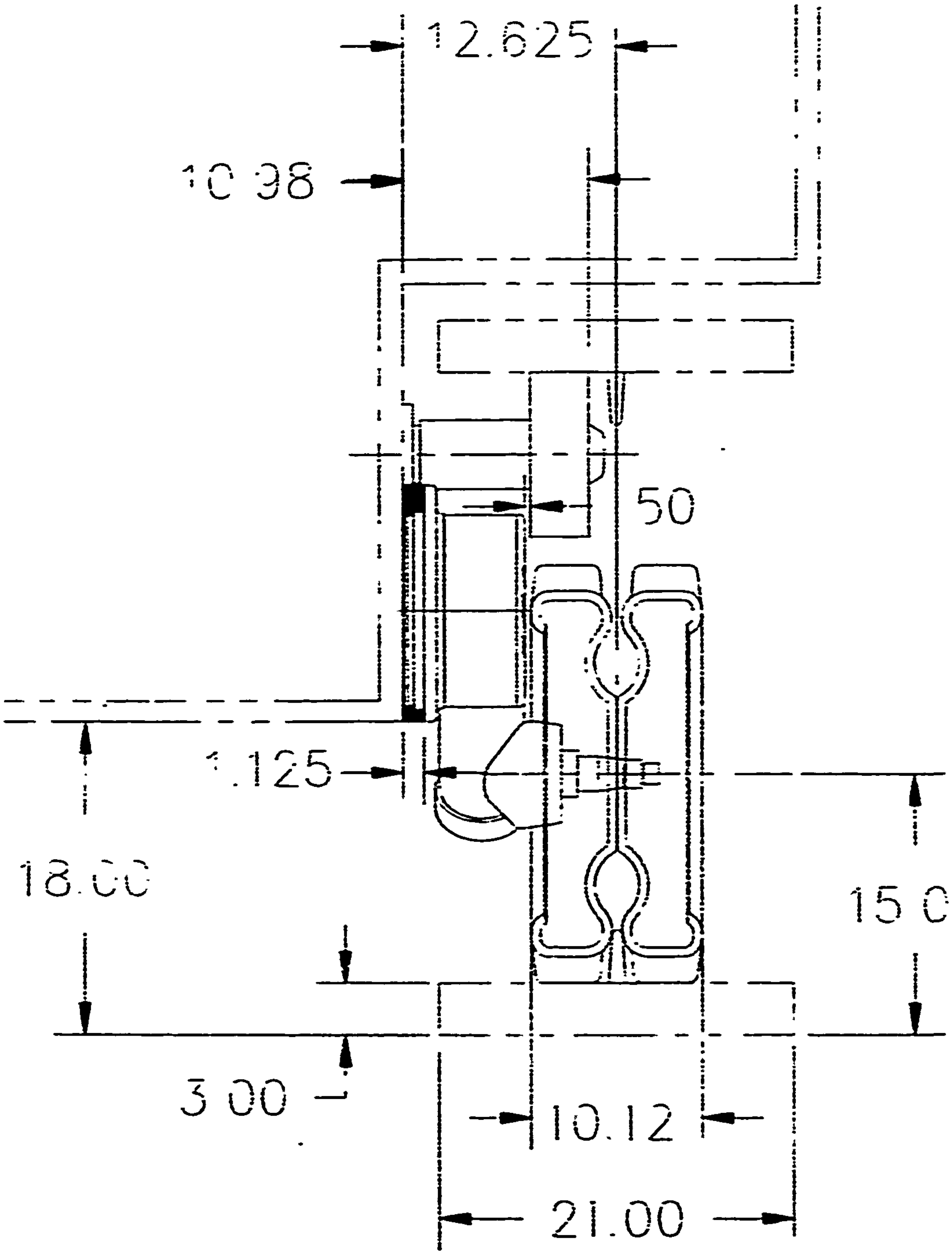


Figure 4

END VIEW

2.3.1.4 Spring curve program.

To determine entrapped gas volumes and pressures, key parameters of the spring geometry were input into a spring curve program developed by Cadillac Gage. These parameters were: roadarm and connecting bar length, piston diameter, crank pin location, the vertical dimension of the road wheel spindle in the static position, and the inches of rebound and jounce travel, and static and jounce wheel loads. This program resolves the kinematics of the unit, shown in Figure 5, and uses this information to calculate the static and jounce pressures at room temperature. It then employs the real gas laws to find the gas density. Assuming adiabatic compression, the density and temperature at jounce pressure are determined. Once the information for the static and jounce positions is complete, the required volume and piston stroke for each position can be calculated. Also, the number of moles of gas required is found. This value is a constant and can be divided by the volume to find the gas density at each position, and then the specific volume. Using entropy chart information stored in the program, the approximate temperature is established for each position, again assuming adiabatic compression. Pressures are calculated, once again by using the real gas laws, and from these the road wheel loads are determined. The program also finds the piston side load, the connecting bar force and the torque at each position. The output for stations 1-6 is given in Appendix C.

2.3.1.4.1 Entrapped gas volume.

The entrapped gas volume is determined by establishing the jounce road wheel load and using the spring geometry to determine the resultant pressure. These values are then used with the real nitrogen gas laws to establish the required gas volume.

The jounce load was specified by Cadillac Gage to be 3.5 g's at all stations for the 6K unit, and 3.0 g's for the upweighted 7.2K unit. The maximum pressure was determined to be 12,500 psi at jounce. The required static gas volume is 53.30 cubic inches.

TSU KINEMATICS

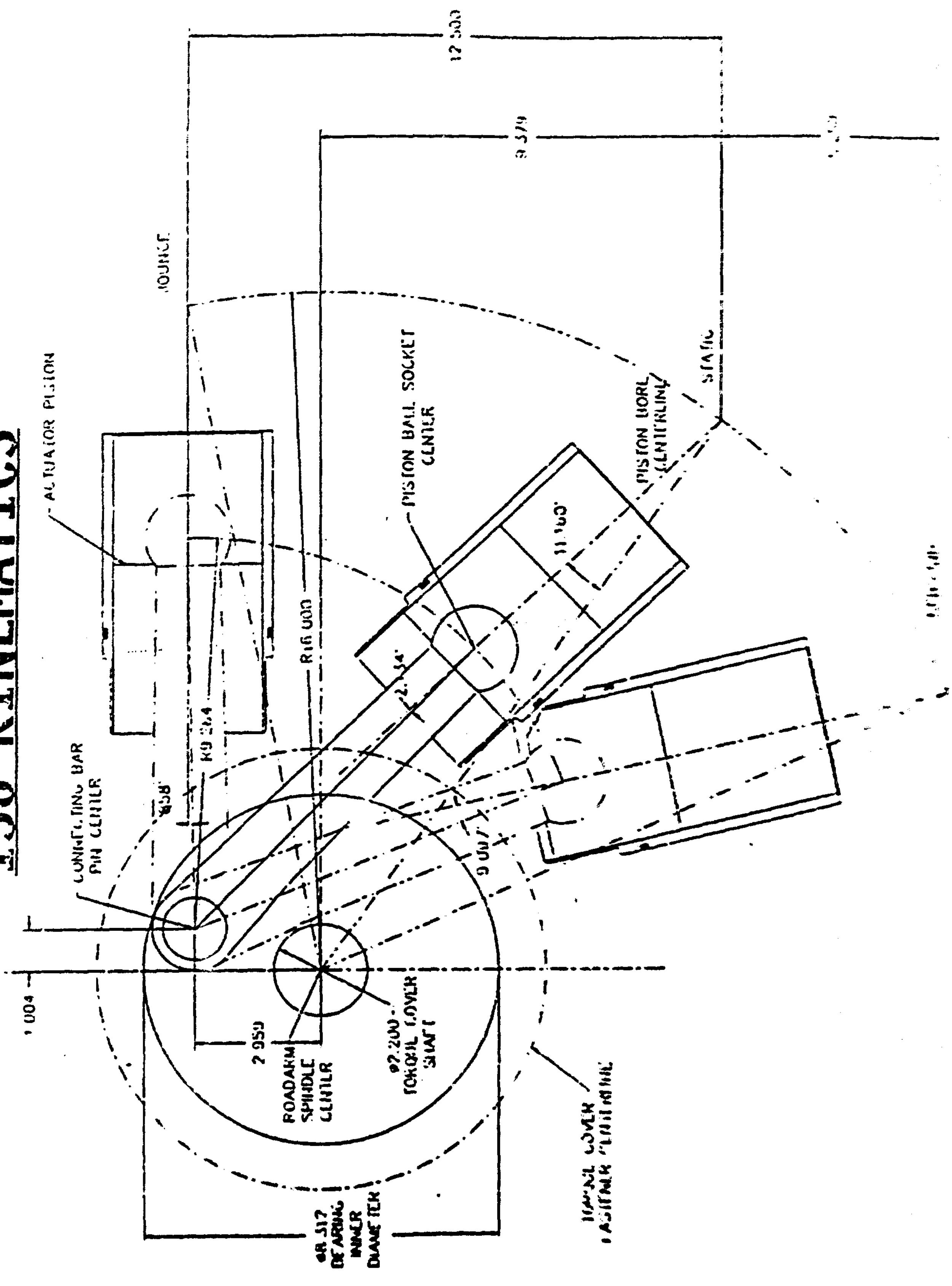


Figure 5

2.3.1.5 Spring sealing system.

A piston seal was sized for the ISU design based on the velocity of the unit and the force seen by the spring system.

This seal consists of a wedge loaded sealing surface acted on by an expansion ring with multiple coil springs to provide an initial preload on the sealing surface. The wedge seal is fabricated from polyimide, which is a very high modulus, high strength plastic. The wedge seal material resists extrusion under high pressure while conforming to irregularities in the mating steel surface, thus forming an effective high pressure seal.

Two Cadillac Gage designed spring loaded seals are used in series on the piston. This configuration forms a buffer zone between the two seals; the first seal is exposed to the full pressure variation while the second is exposed to an almost constant static pressure of 3460 psig. In addition, the first seal acts as a check valve by venting any increase in buffer zone pressure back to the oil chamber when the spring returns to static pressure conditions. This provides minimum leakage and wear of the second seal, and hence, improves gas spring system reliability/durability.

The piston diameter and space claim between piston and cylinder dictated the seal size. Nominal spring preload was calculated to be 1,057 pounds, assuming a 500 psi equivalent spring force to seat the seal at low pressures, plus the additional force needed to size the seal to the bore. Radial load on the seal surfaces due to 3,460 psi pressure was determined to be 5,626 lbs, giving a total load of 6,683 lbs. The total pressure on the dynamic surface is then 5,523 psi. The average velocity of the piston was calculated to be 37.1 feet per minute (fpm). The product of the pressure and velocity is a design constant, the P-V value, used to determine the seal material required. For this ISU design, the P-V value is 204,900 psi-fpm; Vespel, a seal material which has been proven successful up to 250,000 psi-fpm, was chosen for the seal.

2.3.1.6 Spring summary.

The spring system chosen for the 6K ISU has been selected based on the successfully developed and tested designs of the 10K and 4K ISU's. A similarly configured 4K design successfully completed performance testing, repeated 310 hours of durability testing, proof load testing, and a repetition of performance testing during ISU development. Also limited durability and performance t- ts were conducted at various high and low temperatures in an en. onmental chamber. Results of this testing have shown the spring system to be consistently operating within design specifications with negligible leakage.

2.3.2 Damper System.

2.3.2.1 System description.

The components of the damper mechanism consist of a hydromechanical open loop control system and a wet, multiple-friction disc pack installed within the roadarm spindle and activated by the rotating roadarm. The cross-section of the damper is shown in Figure 6. The damper system functions when the roadarm assembly rotates, which forces a cam mounted to the roadarm to drive a piston pump, thereby developing pressure. This pressure, which by v..tue of pump cam cut and relief valve is a function of wheel velocity and position, acts on the damper piston, which then develops an axial clamping force on the friction disc pack. The disc pack's function is to provide a constant slip torque proportional to the applied clamp load, which when rigidly connected to the roadarm, translates into a wheel damping force opposing roadarm rotation. Maximum pressure and pressure rise rate are controlled by the relief valve. When the roadarm returns, the spring-loaded pump and check valve resupply the hydraulic fluid in the control system, and damper springs provide a nominal load on the disc pack for rebound damping. The damper subcomponents are discussed in detail in the following sections, and are shown schematically in Figure 7.

DAMPER

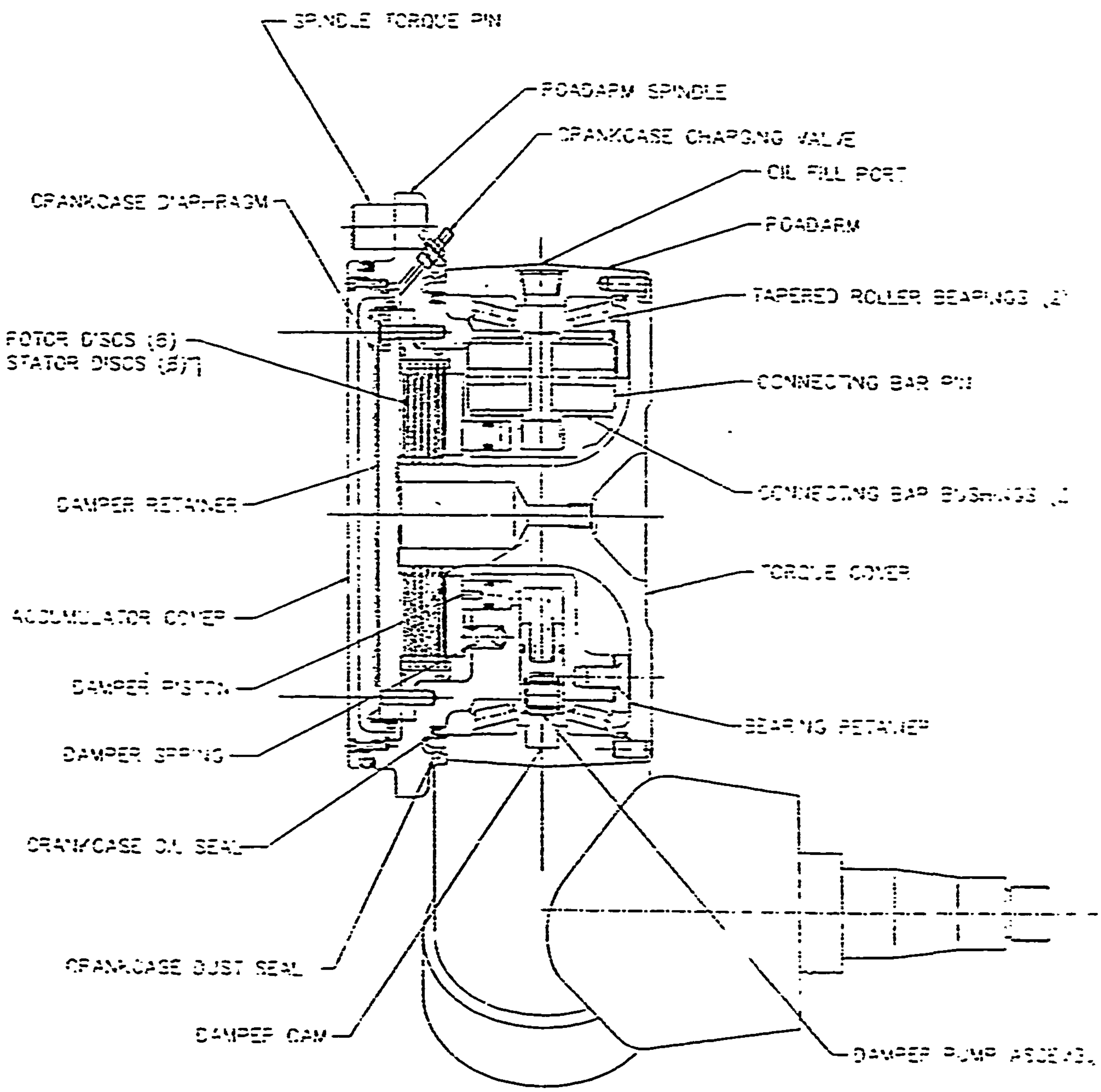


Figure 6

CONTROL SYSTEMS OPERATIONS

DAMPER SCHEMATIC

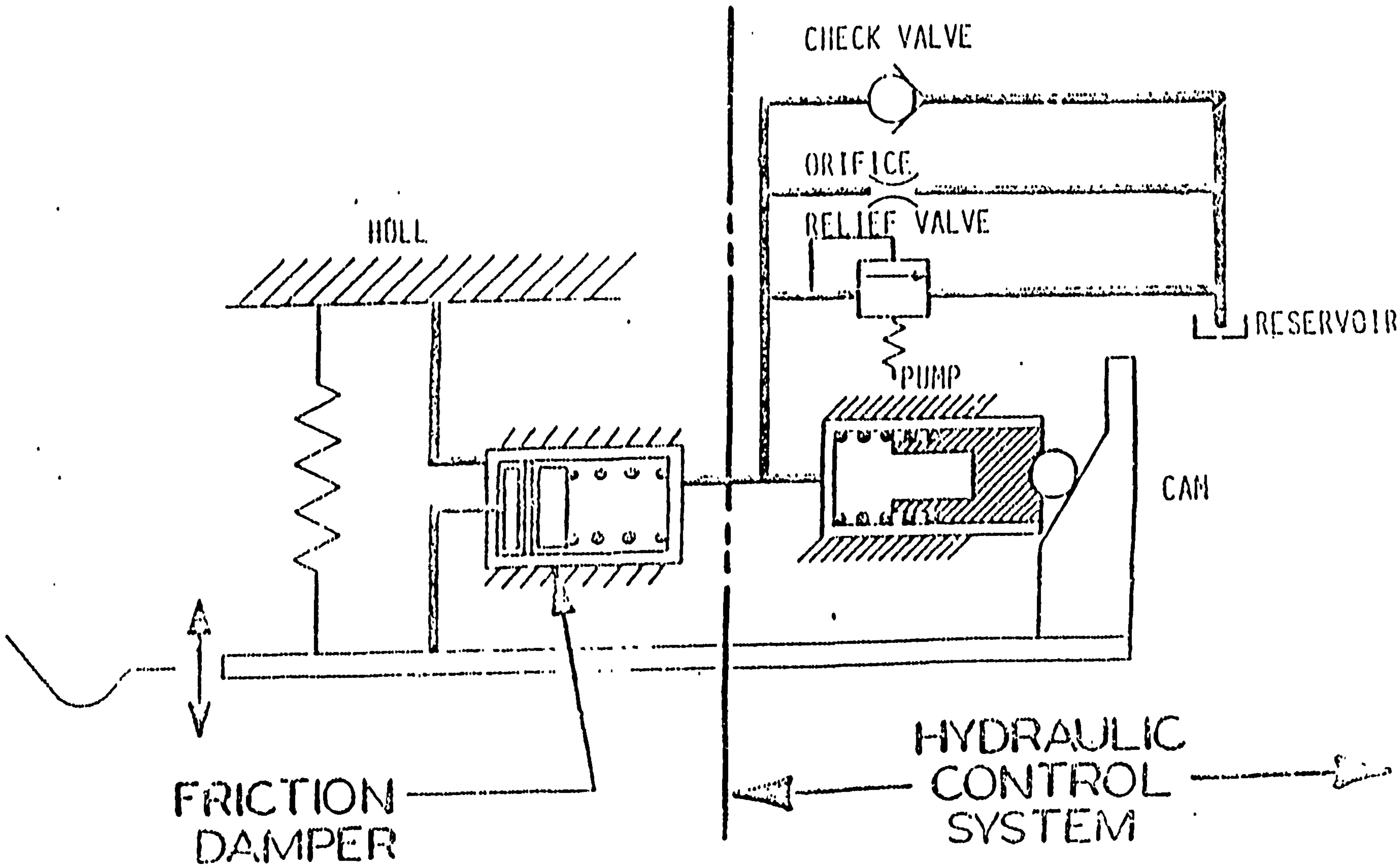


Figure 7

2.3.2.2 Vehicle Dynamics Module

To simulate vehicle ride and shock capability, the Vehicle Dynamics Module (VEHDYN) was employed. VEHDYN predicts the gross motions of a vehicle chassis when negotiating rough terrain or discrete obstacles, and calculates the resulting absorbed power or largest peak accelerations from the vertical accelerations at a specified location in the vehicle. Input includes:

- distances from the sprung mass center of gravity (CG) to the front- and rear-most points on the vehicle, inches
- distance from the CG to the drier's seat, inches
- GVW, pounds
- vehicle pitch moment of inertia, lb-sec^2-in
- longitudinal distances of each wheel center from the CG, inches
- track geometry
- track tension
- suspension spring force - deflection values
- damping rates

Utilizing the static wheel loads and appropriate spring curves from Figure 1, three damping rate combinations were inputted: 1) All stations damped at 0.7 g (4,200 lbs), 2) stations 1, 2 and 6 damped at 0.7 g and stations 3-5 undamped, and 3) stations 1, 2 and 7 variably damped from 0.7 to 1 g and stations 3-5 at 0.7 g. The first and second combinations proved to give the most satisfactory ride quality (see Figure 8). Since the resulting curves were so close, damping on all stations was deemed unnecessary, and the second damping combination was selected for the design. Both

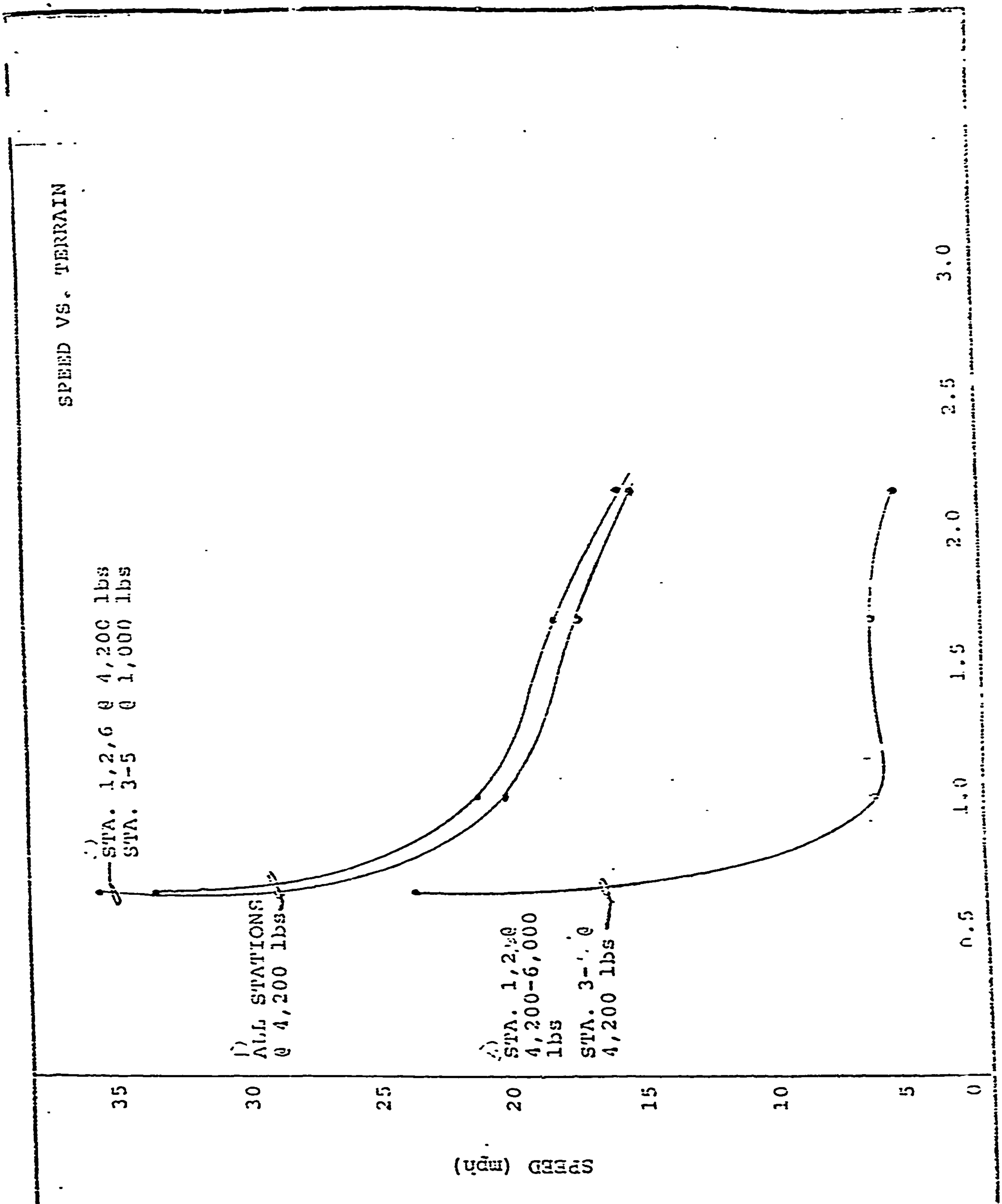


Figure 8

VEHDYN Damping

combinations will be evaluated during field testing to validate these results.

2.3.2.3 Friction disc pack.

The friction disc pack is comprised of seven (7) stator discs alternately placed among six (6) rotor discs. Both are made from medium carbon steel, with the stator discs having paper-based friction material bonded to either side. The stator discs are held stationary through an external involute spline connection to the stator, which is held through dowel pins to the roadarm spindle. The rotor discs are secured to the roadarm through an internal involute spline connection to the torque cover, which is in turn bolted and doweled to the roadarm.

The number of stator and rotor discs required was determined by calculating the number of active surfaces needed to dissipate the energy produced by the amount of torque in the damper, while keeping the energy density per surface at a design limit of 2.0 horsepower per square inch (HP/in^2). This criterion was established through previous ISU testing by Cadillac Gage as the critical value for assured clutch pack performance and life. Twelve active surfaces are required for this design, which gives an average energy density of 1.84 HP/in^2 per surface.

2.3.2.4 Pump assembly.

The pump assembly is one of the key elements in the hydraulic circuit which controls the damper. The function of the pump assembly is to generate pressure which is used to apply a load on the friction disc pack through the damper piston, as well as to provide a flow of oil to the damper system to act as a coolant. The amount of pressure required is dictated by the friction disc load needed to provide a torque output of 80,000 in-lbs, minus the load provided by the damper springs. This pressure was calculated to be 2,865 psi. The oil flow required to the damper system was determined to be 17.20 in^3/sec.

2.3.2.5 Relief value.

The function of the relief valve is to control pressure within the hydraulic control system. The value components consist of a poppet, spring, spring retainer, shims, and retaining ring. An annular clearance around the poppet stem serves as a control system orifice which controls the pressure rise rate within the hydraulic circuit and helps to modulate and stabilize the relief valve. The preloaded coil spring and differential area manufactured into the valve sets the cracking pressure; an 80-lb spring was selected to hold the valve closed up to 2,865 psi of pump pressure. Additional sleeve material has been allotted to allow for a flow-compensated valve should the need arise during testing. Figure 9 shows a cross-section of the valve.

2.3.2.6 Damper springs.

Damper springs, located in the roadarm spindle between the pump assembly and the damper piston, are used to preload the piston, and hence the friction disc pack, for improved damper response and to provide rebound damping. The required preload was determined based on previous ISU applications, where it was observed that a target value of ten percent of the static wheel load was a good maximum value for the wheel load provided by the damper as a result of the springs. For this design, the damper wheel load, then, is 720 lbs; the torque required is 9,330 in-lbs. The spring force needed to provide this torque is 2,680 lbs, which is distributed among nine (9) springs.

2.3.2.7 Check valve.

The check valve's function is to replenish the piston pump hydraulic cavity during return rotation of the roadarm. A spring-loaded poppet design was chosen for this application with the valve seat inserted into the roadarm spindle. A spring load of 0.165 lb is required to accommodate the cracking pressure of 1.5 psi.

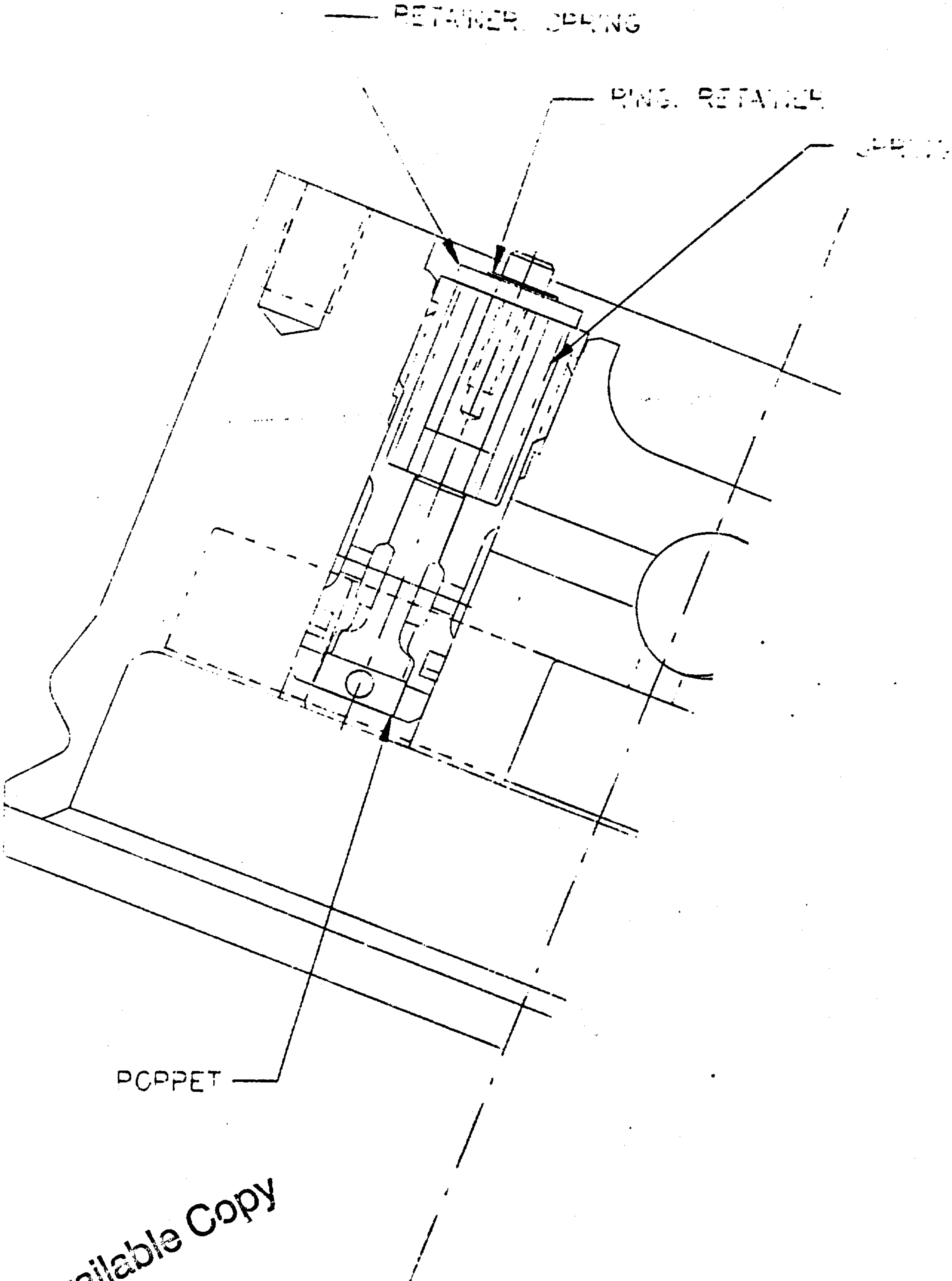


FIGURE 9
RELIEF VALVE ASSEMBLY

2.3.2.8 Crankcase seals.

Two sealing elements are employed to respectively seal in the correct level of damper fluid and pressure and seal out environmental contaminants, thus assuring proper damper system operation. The first, the crankcase oil seal, is a double-lip, spring-energized design. The seal jacket is a Hytrel polymer chosen for its low frictional properties, strength, and wear resistance. The spring energizer material is a stainless steel spring per MIL-S-5059. The crankcase dust seal is a single-lip wiper ring design meeting the requirements of MS28776 and MS33675. This sealing combination has successfully completed rigorous component, unit durability, and field testing on similar ISU designs.

2.3.2.9 Crankcase Diaphragm

The damper components must be bathed in hydraulic fluid to remain operable; however, a certain amount of air must also be maintained so that crankcase pressure doesn't become excessive when volume decreases due to piston movement into the rebound position. A solution to this is a movable diaphragm between the damper retainer and cover. The diaphragm material is a layer of specially-woven fabric impregnated with an elastomer. As the crankcase volume is decreased due to piston movement, the diaphragm is pushed out toward the damper cover, permitting the air volume and crankcase pressures to remain at the required levels.

2.3.2.10 Damper summary.

The complete ISU damper system and its subassemblies are a proven, low risk design that has been thoroughly performance- and durability- tested in both component form and in fully assembled unit configurations. The results of this testing have shown conclusively that the damper system has met the design requirements and is qualified for use in the 6K ISU.

2.3.3 Structure.

The final functional element of the ISU is the structural element. Stress information gained from previous ISU design experience contributed to the design of the major components of this medium ISU. Both classical and finite element techniques were then used to assure the structural integrity of the unit.

2.3.3.1 Load criteria.

All suspension unit components shall have adequate structural integrity to withstand the various load conditions defined below without permanent deformation, long-term durability failures, or extreme rupture, as applicable.

2.3.3.1.1 Maximum load conditions.

	Wheel Load, lbs (g)	
Load Condition	Vertical	Lateral
Static	7,200 (1 g)	0
Jounce	21,600 (3 g)	0
Combined	21,600	±17,300
Max. Lateral		±20,000
Max. Vertical	34,245 (proof)	

2.3.3.2 Stress analysis.

A classical stress analysis was performed for all major components that might be affected by the forces exerted on the ISU assembly during its operation, including a determination of minimum and maximum stresses and fatigue characteristics where deemed necessary. A finite element analysis was completed for the initial roadarm design, roadarm spindle, torque cover, and end cap. These results are given in the following paragraphs and summarized in Table II.

TABLE II

UNIT CRITICAL STRESS SUMMARY

Description	Material	Stress, psi	Classical Analysis	Safety Factor
Roadarm	St 4340, R_c 40-43	79,574	Combined	1.50
Connecting Bar	St 4340, R_c 40-44	128,900	Compressive	1.23
		177,900	Contact	1.29
Actuator Cylinder Sleeve	St 52100, R_c 55-60	147,000	Hoop	1.52
Accumulator End Cap	St 4340, R_c 34-38	111,606	Bending	1.22
Roadarm End Cap	St 4340, R_c 34-38	111,667	Bending	1.20
Torque Cover	St 4340, R_c 36-40	42,300	Torsion	1.29
Damper Stator	St 4340, R_c 38-42	112,400	Tooth Shear	1.21
Damper Stator Pin	Alloy St, R_c 47 min	33,680	Shear	1.86
		30,390	Tension	2.63
Damper Retainer	St 4340, R_c 32-36	41,000	Bending	3.1
Damper Piston	St 4340, R_c 32-36	45,300	Bending	2.8
Stator Disc	St 1035-1085, R_c 20 min	12,835	Tooth Shear	2.0
Rotor Disc	St 1035-1085, R_c 20 min	23,285	Tooth Shear	1.1
Accumulator Cover	AL 7075-T7351	34,656	Bending	1.64

Roadarm, P/N AS700000

The roadarm is forged out of high alloy (4340) steel, heat-treated to 40-43 Rockwell (R_c) hardness. This structure houses the gas/oil spring components. Preliminary hoop stress calculations showed that a 0.38-inch minimum wall thickness was required to withstand a proof pressure of 19,000 psi without exceeding a stress value of 120,000 psi (75 percent of the material yield strength, or a safety factor of 1.3). Finite element analysis was then performed on this design, and the wall thickness was found to be inadequate in certain areas (see Appendix D for the results of this analysis). Using a ratio of the maximum stress found by the finite element analysis to the desired stress limit, a wall thickness of 0.5 inches was determined to be the minimum needed. Classical analysis proved this to be correct; a plot of the calculated fatigue strength falls just below the infinite (10^6 cycle) lifeline on a constant life fatigue diagram.

Roadarm Spindle, P/N AS700001

The cast roadarm spindle, an 8632 steel with a 40-43 R_c hardness, houses the damper components and provides the mounting structure for the roadarm. Stress analysis of the spindle was completed using finite element techniques only. The results obtained showed the stress levels to be acceptable, with the exception of high stress concentrations in the connecting pin bore area. However, it is believed that this is due to the model's assumption of an infinitely rigid connecting pin and a linear load distribution. In actuality, this distribution is nonlinear, but to accurately simulate this would be complex, time-consuming, and beyond the scope of this project. Distributing the load across the width of the pin bore, rather than incorrectly having it concentrated on the bore edge, would reduce the stress to acceptable levels. Also, the model does not take into account

the bushing that is press-fit into the bore, which adds stiffness to the area. See Appendix D for the results of this analysis.

Connecting Bar, P/N AS700044

The connecting bar, 4340 steel at 40-44 R_c hardness, transmits forces resulting from the pressures developed by the gas spring via the connecting bar pin on which it pivots. The maximum compressive stress that the bar experiences was calculated to be 130,000 psi, which results in a safety factor of 1.2. The maximum contact stress, seen where the connecting bar and ball meet, was determined to be 190,000 psi, also giving a 1.2 factor of safety.

Connecting Bar Pin, P/N AS700021

This pin functions by transmitting forces from the rod or piston assembly back to the spindle assembly. The pin is manufactured from 52100 steel, hardened to a 58-63 R_c. At proof pressure, the stress on the pin surface due to fully reversed bending was determined to be 180,584 psi, which gives a safety factor of 1.25. The maximum shear stress at the pin's center was calculated to be 73,414 psi, resulting in a 1.75 safety factor. For endurance in fully-reversed bending, the stress at the maximum operating pressure was found to be 118,792 psi, with a 1.18 safety factor. Shear stress was calculated at 48,293 psi, giving a safety factor of 1.65.

Connecting Bar Bearing, P/N AS700042

The two connecting bar bearings are self-lubricating, fabric-lined bearings that transmit loads seen by the connecting bar pin to the spindle and provide ease of pin movement by reducing friction. Maximum load at proof on each bearing is 91,400 lbs.

Main Bearing, P/N AS700046

The roadarm is supported on the roadarm spindle by two 52100 steel, full-compliment tapered roller bearings. These bearings provide the load-carrying capacity for forces produced by the damper, spring, and induced road wheel loads, as well as minimizing the friction produced by angular movement of the roadarm. A load analysis shows a maximum radial load of 149,750 lbs on each bearing; they were sized by the manufacturer (Kaydon Corporation) to withstand the loads incurred and to fit within the allowable space between roadarm bore and spindle.

Actuator Cylinder Sleeve, P/N AS700037

The translating motion of the piston through the actuator cylinder is a result of the angular motion of the roadarm. The cylinder sleeve through which the piston travels is mainly subject to hoop stress resulting from pressures in the gas spring. The allowable thickness of the sleeve results in a stress of 147,000 psi. The chosen material is 52100 steel, hardened to 55-60 R_C, which has a yield strength of 224,000 psi. This gives a safety factor of 1.52 for the sleeve.

Accumulator and Roadarm End Cap, P/N AS700005 and P/N AS700004

The accumulator and roadarm end caps are secured to the ends of the gas and actuator cylinders, respectively, with the use of buttress threads. Both experience bending stresses resulting from cylinder pressures. The maximum deflection seen by the accumulator end cap is 0.0017 inches; the maximum bending stress seen, at proof pressure, is 111,606 psi, giving a 1.22 safety factor. The roadarm end cap has a maximum deflection at the neutral axis of 0.0036 inches and a maximum bending stress of 111,667 psi giving a 1.2 factor of safety. The fatigue limits for both end caps fell below the infinite lifeline on a constant life fatigue diagram for the material, which is 4340 steel at 34-

38 R_c hardness. Also, a finite element analysis was performed on the initial design; results appear in Appendix D.

Torque Cover. P/N AS700002

The torque cover conducts a damping force developed by the rotors through a spline, machined on an integral shaft, which tapers into a flat cover that mounts to the front of the roadarm assembly. The material is forged 4340 steel, hardened to 36-40 R_c. The maximum applied load is in torsion and is equal to 80,000 in-lbs. This produces a maximum shear stress of 42,300 psi on the shaft. The allowable endurance strength in shear is 54,720 psi, resulting in a 1.29 safety factor.

Torque Cover Pin. P/N AS700315

The cover pin reacts to the force developed by the friction damper and cover which is transmitted to the roadarm. There are three pins at the cover/roadarm interface which operate in a shear mode. The developed load per pin is 5,079 lbs. The rated single shear strength for each pin is 9,710 lbs, giving a 1.91 safety factor.

Damper Stator. P/N AS700023

The damper stator transmits friction damper-developed force through the stator damper splines to the roadarm spindle. The maximum load applied to the O.D. is 23,600 lbs, which results in a tooth shear stress of 112,400 psi. The stator material is 4340 steel with a hardness range of 38-42 R_c, for a 1.21 safety factor.

Damper Stator Pin. P/N MS16555-41

The damper stator pin restrains motion between the stator damper and the roadarm spindle. The shear stress for each of

three pins is 33,680 psi, which results in a safety factor of 1.86. The tensile stress for each pin is 30,390 psi; using a material fatigue strength of 80,000 psi, the resulting safety factor is 2.63.

Damper Retainer, P/N AS700022

The damper retainer locates and constrains the friction damper package when the piston damper applies a load. The maximum load applied to the retainer was determined to be 22,000 lbs, which results in a bending stress of 41,000 psi and a deflection of 0.011 inches. The material is 4340 steel, heat-treated to 32-36 R_C hardness, which has a yield strength of 128,000 psi. This gives a safety factor of 3.1.

Damper Piston, P/N AS700019

The damper piston applies an axial load to the friction damper. The piston develops a maximum stress of 45,300 psi and a deflection of 0.0026 inches. The resulting safety factor is 2.8.

Stator Disc, P/N AS700017

The stator disc is externally splined with friction material bonded to either side. These discs transmit frictional force from the rotor discs to the stator damper pinned to the spindle. The minimum recommended plate thickness is 0.030 inches, which results in a spline tooth shear of 12,835 psi. The material is steel alloy, 1035-1080, hardened to a minimum of 20 R_C. The safety factor was calculated to be 2.0.

Rotor Disc, P/N AS700018

The rotor discs are internally-splined discs that are located alternately between the stator discs and fabricated from the same

low-carbon steel. The tooth shear developed is 23,285 psi, which results in a safety factor of 1 1.

Damper Accumulator Cover, P/N AS700304

The accumulator cover, manufactured out of 7075-T7351 aluminum, encloses the damper diaphragm and components. Due to the proof pressure of 200 psi seen by the crankcase, it experiences a maximum bending stress at the outer diameter of 34,656 psi. The safety factor is 1.54.

Damper Check Valve Assembly

The check valve assembly permits one way flow for the system hydraulic fluid and then seals completely to allow pressure buildup in the hydraulic control system. Maximum loads applied to this system result in impact and torsional stresses.

Check Valve Spring, P/N AS700039

The check valve spring is used to position the poppet against the check valve seat. The torsional stress at the solid height position is 14,820 psi which results in a safety factor of 10.39.

2.4 Environmental Effects

This hydropneumatic suspension design has been developed to minimize the environmental impact on the integrity of the ISU system.

2.4.1 Thermal.

A gas spring's operational characteristics are affected by both ambient temperatures while not in operation and by an increase in the system temperature from the damper during dynamic operation of the vehicle.

2.4.1.1 Ambient.

In accordance with the laws of thermodynamics, temperature has a significant impact on the nitrogen gas pressure in the spring system. The gas pressure will increase or decrease as temperature increases or decreases, which will vary the force opposing the road wheel load. The result will be a variation in the vehicle height due to changes in temperature. Optimizing the spring system geometry will minimize this effect, but not eliminate it. The results of a temperature sensitivity analysis performed for this design shows that a ± 60 °F change will alter the vehicle height by ± 1 inch. This analysis appears in Appendix E.

2.4.1.2 Operational.

Dynamic operation of the vehicle also impacts the nitrogen gas spring due to the temperature increases developed in the damper section. Experience has shown these effects to be minimal. Due to the large size of the frictional damper and efficient heat transfer into the vehicle hull, temperature effects due to dynamic operation of the unit have been minimized.

2.4.2 Corrosion.

All necessary precautions will be followed to insure the protection of the ISU from corrosion. These precautions include the appropriate material selection, surface treatment, corrosion protection, priming and painting as defined in standards:

MIL-HDBK-132A	Protective Finishes for Metal and Wood Surfaces
MIL-STD-171D	Finishing of Metal and Wood Surfaces
MIL-STD-193K	Painting Procedures and Marking for Vehicles

Further, components of the ISU exposed to saltwater are finished to minimize galvanic corrosion based on the guidance provided in Table I of MIL-STD-171 to ensure the compatibility of mating components made of dissimilar metals.

Corrosion protection techniques are detailed in a Corrosion Prevention and Material Deterioration Report presented in Appendix F.

2.5 Ballistic Protection

To establish the armor thickness required for protection from a 7.62-mm armor-piercing shell at 0° obliquity, a computer program was used that was written by Cadillac Gage based on published information from Johns Hopkins University and other sources, including the Army materials and Mechanics Research Center. It was determined that a 0.5-inch armor thickness was needed. Appendix G shows the computer results and gives the source for each set of calculated values.

2.6 Suspension Interface

The GFE AAV-7A1 will first be modified to provide an appropriate mounting surface for the ISU's. This includes covering and sealing the previous mounting surfaces for the torsion bar suspension system. An adaptive mounting plate will be welded to the hull to accommodate each of the ISU's on the vehicle hull. Modifications to the hull will also include the addition of two (2) support rollers per side to separate the track from the ISU's.

The suspension units will be installed on the GFE AAV-7A1 with the use of a large, 11.50-in. pilot diameter, an anti-rotation dowel pin, and secured through the adapter plate and into the hull with nine (9) hex head fasteners. The large pilot diameter and dowel pin allow for quick and simple installation and alignment of the unit. Appropriately-protected fasteners secure the unit to the assumed aluminum hull with the use of Rosan inserts.

2.7 Maintenance

The ISU is designed to be easy to maintain by virtue of positioning the damper charge valve, gas spring charge valve, and crankcase oil filler

plugs in locations that are accessible with the road wheel installed on the unit.

A level measuring device is incorporated on the torque cover to allow for examination of the fluid level inside the crankcase which also acts as the damper oil reservoir. The damper fluid need not be changed over the life of the ISU, and will rarely, if ever, require additional oil except in the case of unit damage. The crankcase pressure can be checked using a tire pressure gage on the damper charge valve located on the top of the crankcase. A relief valve, incorporated as a precaution against overpressure in the crankcase, is self-cleaning and should not require servicing.

The gas spring is easily maintained and is designed for simple service at a six-month interval. Inspection or adjustment of the spring pre-charge is easily accomplished through the charging valve located on the top of the roadarm near the end cap. Any need to adjust the gas spring may be determined by changes in the vehicle height and attitude.

2.8 Unit Weight

As a result of design considerations to minimize weight while assuring structural integrity, the total weight value for the unit, including fluid, road wheel spindle, and hub assembly, is estimated to be 281.3 pounds, ±4 percent. This estimation takes into consideration tolerances required for forgings, castings, and machining; during fabrication and testing, every effort will be made to reduce weight wherever possible. A summary of ISU component weights is given in Table III.

3.0 SYSTEM CHARACTERISTICS

The hydropneumatic suspension system design presented herein incorporates a vehicle ground clearance of 18 inches, which exceeds the Request for Proposal requirement, and a road-wheel diameter of 24 inches, as required.

Table III

Unit Weight Summary

Description	Part No.	Weight, Lbs
Spindle, Roadarm	AS700001	53.4
Spindle, Wheel, Vehicle	AS700015	7.56
Pin, Spindle, Torque	AS700314	0.39
Piston, Actuator	AS700009	3.68
Cap, Piston, Actuator	AS700036	0.57
Bearing, Conn Bar	AS700042	0.27
Ring, Retainer, Conn Bar	AS700038	0.08
Ball, Conn Bar	AS700045	0.98
Rod, Conn Bar	AS700044	3.71
Ring, Glyd	AS700013	0.04
Seal, Piston, Actuator	AS700012	0.02
Ring, Expansion, Actuator Piston	AS700028	0.30
Spring, Comp. Seal	AS700069	0.05
Ring, Spacer	AS700035	0.17
Spacer, Seal	AS700033	0.43
Valve, Oil Fill	P42503	1.60
Diaphragm, Accumulator	AS700318	0.20
Cover, Accumulator	AS700304	3.12
End Cap, Accumulator	AS700005	3.67
End Cap, Roadarm	AS700004	5.32
Screw, Valve	DP41181	0.07
Stem, Valve	DP41182	0.02
Plug	MS49005-8	0.02
Screw, Cap, Hex Head	MS90728-9	0.02
Screw, Cap, Hex Head	MS90728-5	0.06
Screw, Flat Head, Hex Socket	39895	0.08
Cam, Pump	AS700049	1.39
Screw, Cap, Hex Head	MS90728-9	0.01
Roadarm	AS700000	98.1
Cover, Torque	AS700002	19.22
Plug and Bleeder	AN-814-4	0.02
Plug, Hollow Hex, O-Ring	4HP50N	0.02
Pin, Drive	AS700315	0.35
Retainer, Bearing	AS700014	2.24
Shim, Bearing Retainer	AS700048	0.25
Piston, Damper	AS700019	3.01
Ball, Check Valve	MS19059-4826	0.10
Spring Guide, Check Valve	AS700026	0.05
Seat, Check Valve	AS700025	0.08
Spring, Check Valve	AS700039	0.02
Seal, Crankcase Oil	DP41198-2	0.01
Seal, Crankcase Dust	DP41199	0.01
Pin, Connecting Bar	AS700021	1.58
Disc, Stator Assembly	AS700017	1.69
Disc, Rotor	AS700018	5.51
Race, Inner	AS700311	0.05

Table III

Unit Weight Summary

(Continued)

Description	Part No.	Weight, Lbs
Race, Outer	AS700312	0.05
Piston, Pump	AS700027	0.24
Bearing, Needle	AS700047	0.10
Bearing, Main	AS700046	12.70
Spring, Pump	AS700041	0.02
Retainer, Damper	AS700022	7.21
Stator, Damper	AS700023	2.44
Spool, Relief Valve	AS700029	0.07
Sleeve, Relief Valve	AS700031	0.36
Spring, Relief Valve	AS700040	0.02
Spring Guide, Relief Valve	AS700030	0.01
Screw, Cap, Hex Head	MS90727-32	0.05
Sleeve, Actuator Cylinder	AS700037	4.98
Spacer, Actuator Cylinder	AS700034	0.90
Cover, Charge Valve	AS700309	0.43
Screw, Socket Head	MS16997-47	0.01
Spring, Damper	AS700316	0.35
Plug, Drain	AS700097	0.02
Pin, Stator	MS16555-41	0.05
Hub Assembly, Road Wheel		24.84
Fluid		7.00
Total Unit Weight		281.3

A calculation of the vehicle natural frequencies resulted in a bounce frequency of 1.009 Hertz and a pitch frequency of 0.882 Hertz. This calculation appears in Appendix H.

The ISU is designed to operate as a 3.5 g road wheel load spring, providing 12.5 inches of vertical road wheel travel in jounce and 5 inches of travel in rebound. The unit will withstand a vertical load at the track of 34,245 pounds, a side load on the track of 20,000 pounds, and a simultaneous load of 17,300 pounds to the side with 21,600 pounds vertically.

3.1 System Weight

The hydropneumatic suspension system weight includes the support rollers, adaptive mounting plates, jounce stops, torsion bar hole plugs, and the 12 ISUs. The total estimated system weight is 3,717.8 pounds. A system weight summary is tabulated in Table IV.

Table IV
System Weight Summary

Description	Quantity	Weight, lbs
ISU (w/bolts)	12	3389.7
Support Rollers	4	128.0
Mounting Brackets	12	144.0
Jounce Stops	4	32.0
Torsion Bar Hole Plugs	12	24.1
Total System Weight		3717.8